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The Use of Screening Tests in Spacecraft Lubricant Evaluation

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A lubricant screening test fixture has been devised to obtain lubricant performance data in a timely manner. This fixture has been used to perform short-term tests on potential lubricants for several spacecraft applications. The results of these tests have saved time by producing qualitative performance rankings of lubricant selections prior to life testing. To date, this test fixture has been used to test lubricants for three particular applications: an oscillating scanner bearing, a harmonic drive actuator mechanism, and a reaction wheel support bearing. The qualitative results from these tests have been verified by life test results and have provided insight into the function of various antiwear additives.

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PREFACE

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I. INTRODUCTION

Because of the stringent conditions placed on spacecraft lubricants and the lack of specific performance information, ground-based testing of lubricants is a major factor in the design of spacecraft mechanisms and in the process of selecting suitable lubricants. In most cases, the approach taken when testing lubricants for space qualification is to perform system-level life tests on actual flight hardware, or to perform simulation life tests that attempt to duplicate the conditions of flight system operation. This approach usually produces useful results but is limited by the length of time necessary to obtain data and by the relatively high cost of testing. In most cases, only one or two candidate lubricants can be tested, and their selection is usually based on past experience and not on reliable test information. There is a need for a test procedure that can rapidly screen potential candidate lubricants for applications before system-level life tests are performed.

In the course of our work, we have been confronted by this need on several occasions. In response, we have developed a test fixture and a test procedure that can determine relative lubricant performances before candidate lubricants are committed to an application life test.[1] The test fixture utilizes low cost components and operates under accelerated conditions to reduce the time for lubricant evaluations. The test capability of this fixture enables us to rank lubricants for suitability for application in life testing. In particular, we have used this test approach to select lubricants for the following application tests: an oscillating scanner bearing, a harmonic drive support bearing, and, recently, for a reaction wheel support bearing. With regard to the oscillating scanner and the harmonic drive, the

results of the screening tests have been verified by life
tests. In these situations, the results from the life tests
have produced a relative lubricant ranking that is identical
to the ranking determined by the lubricant screening test
facility.

II. EXPERIMENTAL

The approach developed to evaluate spacecraft lubricants is centered around two different types of lubricant tests. The first type of test utilizes a lubricant screening test fixture, to perform short-term accelerated bearing tests on a number of lubricant selections. The results of this test provide a generalized performance ranking of the selections and indicate the suitability of the lubricants with respect to the intended application. The next type of test is a life test or a simulated life test, to evaluate the suitable lubricants under realistic operating conditions. In both types of tests, posttest surface analyses are performed on the test specimens in order to understand the test results.

A. SCREENING TEST BEARINGS

The bearings used for the screening tests are INA GT-1 thrust bearings. The primary reasons for selecting these bearings was their low cost and availability. These bearings have a complement of 12 balls and a raceway diameter of 0.900 in. The balls, like the races, are made of 52100 steel and have a diameter of 0.187 in. For the screening tests, we changed the configuration in which the bearings were run. Instead of using grooved raceways for both the top and bottom of the bearing, the flat side of the bottom (rotating race) was polished to a 0.25 μm finish and used in place of the raceway. This change in configuration was done so the bearing could be run with a controlled amount of misalignment. The misalignment accelerates the wear process. (The implications of this misalignment will be described later.) In addition, a flat contacting surface operates at higher Hertzian stress than the raceway, which guarantees that more severe tribological conditions will exist on the flat. This situation is crucial to the success of the test because the flat is far easier to analyze than the grooved racetrack.

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B. TEST FIXTURES

The fixture that was developed to perform the screening tests is represented in Figure 1. Based on an earlier design,[2] the fixture was changed significantly in order to enhance its capabilities. Similar types of fixtures, based on the earlier design, have been reported in the literature.[3] The fixture consists of a lower housing, which contains the support bearings for the drive shaft, and an upper housing, which is designed to contain the test bearings. On the top of the fixture is a plate that is used to load the test bearings. This plate is free to slide up and down along four studs that are threaded into the top of the upper housing. The test load is applied to this plate with a set of springs (one for each stud) that are compressed along the studs and against the load plate. This arrangement, in turn, pushes the bottom of the load plate against the stationary race of the test bearing. To determine the applied load, a load cell is placed between the load plate and the stationary race of the bearing. The load cell can measure loads from 0 to 100 lb and has a resolution of 0.1 lb.

The thrust bearings used for the screening tests are mounted in a housing located in the bottom of the load plate. During a test, these bearings are situated such that the stationary (top) race is mounted in the detachable housing and the rotating flat is on a shoulder of the drive shaft. The fixture was designed to incorporate the detachable housing to vary the degree of ball skidding in the test bearing. As described earlier, the ball skidding is produced by mounting the upper race of the thrust bearing slightly off-center from the rotational center of the rotating flat. The resulting misalignment between the top and bottom races causes balls to skid as well as roll during operation. This type of motion imposes severe stress on the bearing and the lubricant because
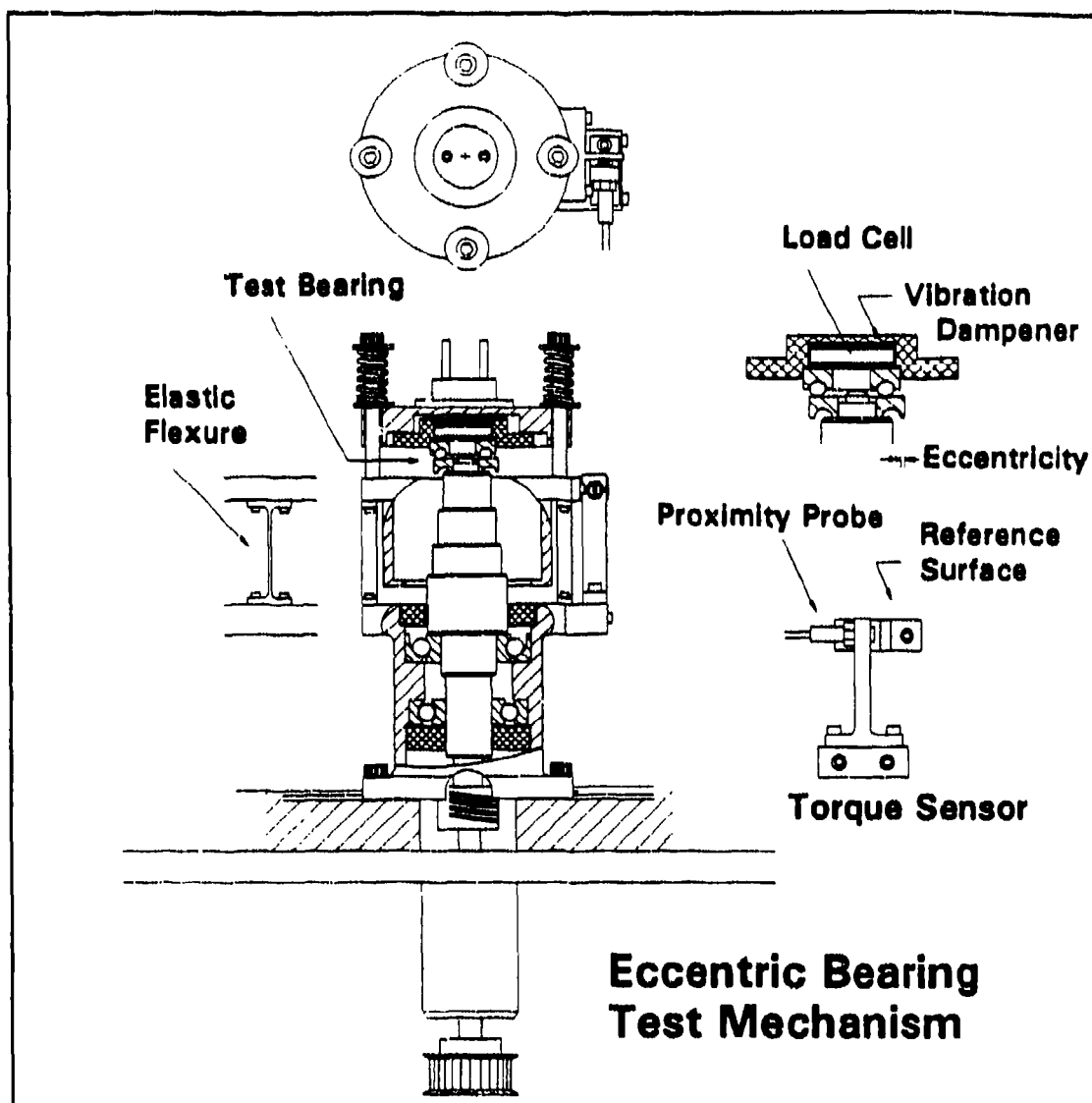
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Figure 1.   Lubricant screening test apparatus.

it forces the bearing to operate in a boundary lubrication regime. Consequently, test lives are greatly reduced in comparison to typical ball bearings. In addition, the eccentric operation of the bearing produces a relatively wide wear zone on the flat disk. This is beneficial because it provides a large area in which to conduct posttest surface analysis. The amount of eccentricity in the test bearing is variable and can be adjusted from dead center to 0.125 in.

To measure the reaction torque of the bearing, the upper housing is connected to the lower portion of the test fixture through a set of aluminum flexures. The flexures allow the upper portion of the test fixture to rotate slightly with respect to the bottom housing when a torque is applied to the housing of the test bearing. The reaction torque of the bearing is thus determined by measuring the amount of rotation of the upper housing with respect to the stationary, lower housing. This measurement is accomplished by mounting an inductive proximity sensor to the lower housing and measuring the change in displacement between it and a target plate mounted on the upper housing. The proximity sensor is calibrated by applying known torques to the upper housing and measuring its stiffness coefficient, which is then used to convert measured displacements into bearing reaction torques.

During the tests, the reaction torque of the test bearing is monitored continuously with a data-acquisition computer. A custom Fortran program is used to acquire the data and store it to a disk file. The data are acquired at a relatively high acquisition frequency (100 Hz) and are then time compressed by a data averaging routine. The routine acquires 100 Hz data in 10 s intervals and computes a mean value for each interval. The mean values are accumulated in a data array and are stored periodically to a data file. The averaging routine allows the reaction torque of the test bearings to be monitored

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continuously throughout the test without storing an excessive amount of data.

To simulate the space environment, the test fixture operates in a vacuum chamber. The chamber is pumped by a 360 1/s turbo-molecular pump and reaches a baseline pressure of 10^{-8} torr. In addition, the chamber has the capability of being sealed off completely and back-filled with gases (e.g., He for certain reaction wheel bearings). Since the drive motor is external to the vacuum chamber, a rotary (Ferro-Fluidics) feed-through is used to transmit rotary motion to the drive shaft of the fixture.

In addition to testing thrust bearings, several different sizes of angular contact bearings can be tested with only minor changes in the upper bearing housing and load plate. Recently, the test fixture was modified to perform pin-on-disk tests. This modification allows data to be obtained on friction and wear of different materials and surfaces.

An accelerated life-simulation test fixture, referred to as the boundary lubrication test fixture, was used to verify some of the results from the screening test. This fixture utilizes modified bearing components as test specimens and operates at speeds and pressures that are very close to those experienced by the bearings in many low-speed oscillatory mechanisms. This test fixture operates in a linear oscillatory manner. The performances of the lubricants are assessed by measuring the wear rate of the test specimens during the testing. This is accomplished with capacitive displacement probes that are mounted directly above the specimens. Figure 2 shows the boundary lubrication test fixture's bearing and sensor configuration in detail.

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Probe Assembly and Raceway Retainers

- Load Spring
- Micrometer
- Cap. Probe
- Positioner

TOP V-BLOCK

BOTTOM V-BLOCK

Wear Samples

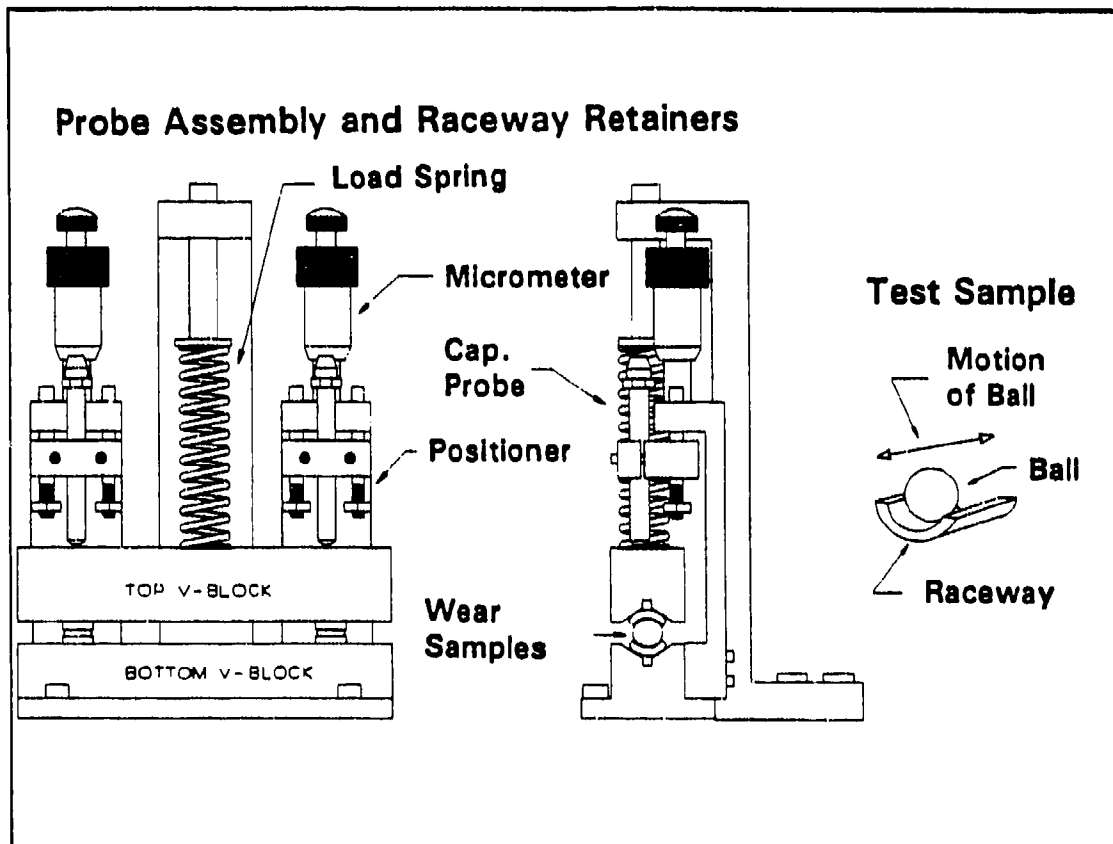**Test Sample**

- Motion of Ball
- Ball
- Raceway

Figure 2.   Boundary lubrication test fixture, bearing, and sensor assembly.

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III. RESULTS AND DISCUSSION

A.   CASE 1 - OSCILLATING SCANNER BEARINGS

In this particular test program, the goal was to find a suitable replacement lubricant for a pair of R2 bearings that are used to support the shaft of an oscillating optical scanner. The original lubricant used in this application was G.E. Versilube F-50, a chloroarylalkylsiloxane (CAS) oil. This oil has a very low vapor pressure and excellent low temperature properties, but it does not function well under boundary lubrication conditions. In this application, the oil degraded very quickly, which resulted in substantial bearing wear and reduced life. The goal of the screening tests was to find a replacement oil that had vapor pressure characteristics and a pour point comparable to the CAS oil as well as enhanced boundary wear characteristics. Once identified, the oil was to be tested in a simulated life test to qualify it as a space lubricant.

After preliminary investigation, substitute oils were chosen for the tests. These oils, along with some of their physical properties, appear in Table 1. The CAS oil was included in the testing to obtain a performance baseline for comparison. The perfluoropolyalkylether (PFPE) oil, Brayco 815Z, was selected on the basis of its low vapor pressure and viscosity properties. The poly-alpha-olfin (PAO) oil, NYE 188B, differs from the other oils because it is a synthetic hydrocarbon oil. It was chosen because of its excellent physical properties and because it could be formulated with the antiwear additive tricresylphosphate (TCP). The inclusion of an antiwear additive in the oil was considered to be essential because the bearings operate in a boundary lubrication regime.

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Table 1. Oils Tested in Scanner Bearing Case Study

Property	CAS	PFPE	PAO

Viscosity, cS
    -40 °C
    40 °C
    100 °C
	640
	52
	16
		2600
		129
		40
			--- 	
			107
			14.5

Viscosity index	--	350	145
Pour point (°C)	-73	-73	-55
Specific gravity	1.045	1.866	--

Table 2. Oils Tested in Harmonic Drive Case Study

Oil	Additive
NPT-4	None
Pennzane SHF-2000	None
Pennzane SHF-2000	Pbnp (5%)
Pennzane SHF-2000	TCP (1%)

In this set of tests, the earlier configuration of the test fixture was used.[1] This configuration utilized a thrust bearing of the same dimensions as the INA GT1 but incorporated a custom 440C disk and 440C grade 10 balls. The test specimens were run in a vacuum environment at a speed of 1750 rpm, and the load ranged between 20 and 50 lb (229-310 ksi peak contact stress). The motor current of the test fixture was measured to get an estimate of the reaction torque of the bearing. Bearing failure was defined to occur when the motor current reached a level 1.5x the starting current.

The wear lives of the screening tests are shown in Figure 3. In this figure, the wear lives of the oils have been normalized with respect to the worst performer. As the figure indicates, the PAO oil had the longest life under these conditions. The primary reason for this result appears to be the superior boundary protection provided by the antiwear additive (TCP) in the PAO oil. In the absence of antiwear additives, the CAS and PFPE oils could not protect adequately against direct metal-to-metal contact in the bearing. In addition, both the CAS and the PFPE oils decomposed under use. The CAS oil was the most reactive, forming a hard "sand-like" grit, which caused significant wear and subsequent failure after only a short period of operation.

After these tests were concluded, the same three oils were tested in a simulated life test of flight grade bearings under conditions that mimicked the operational conditions on orbit. The results of the life testing appear in Figure 4. It is obvious that the PAO oil outperformed the other oils by a wide margin. Both the CAS and PFPE oils exhibited the same failure mechanism in this test as they had in the screening tests. The absence of an antiwear additive, coupled with lubricant decomposition, was the most likely cause of the early bearing failures. In contrast, the PAO lubricated
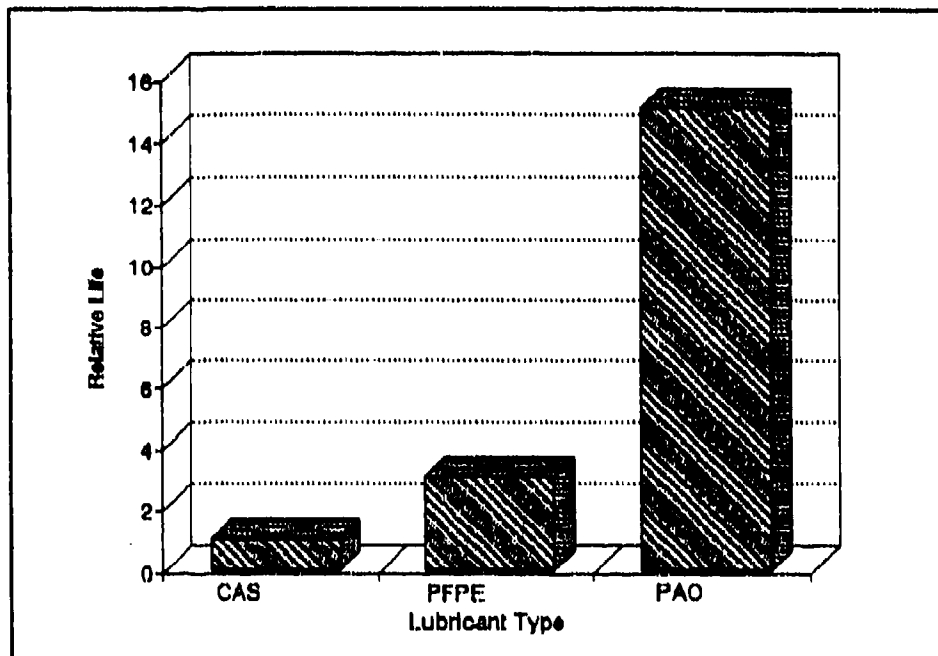
17

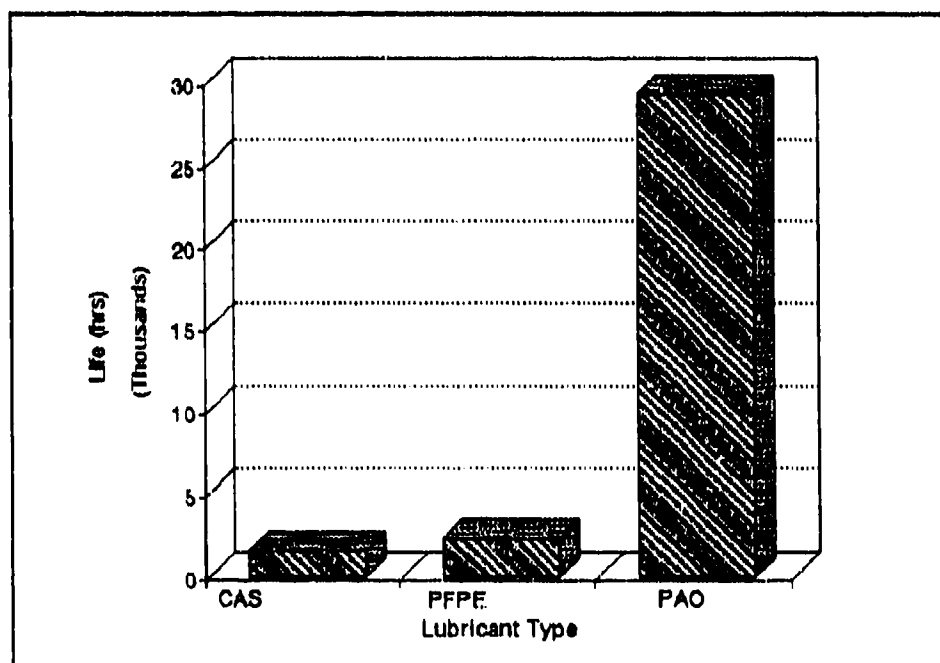Figure 3. Screening test results (scanner mechanism).

Figure 4. Life test results (scanner mechanism).

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bearings operated over 3.5 years and did not display abnormal wear upon removal. The bearings still contained an adequate amount of lubricant, and the lubricant did not show any signs of degradation. More details on these test results can be obtained from the literature.[4]

B.   CASE 2 - HARMONIC DRIVE SUPPORT BEARINGS

Another lubricant evaluation program involved testing lubricants for a harmonic drive actuator mechanism. The primary goal of this program was to obtain comparative performance data for a frequently used neopentylester spacecraft oil (NPT-4) and a new synthetic hydrocarbon oil (Pennzane SHF-2000), a multiply-alkylated cyclopentane (MAC) that has some outstanding physical properties. An additional goal was to gain a better understanding of the role of wear additives in oils and the mechanisms by which they provide protection. The additives of interest were TCP and lead naphthenate (Pbnp).

To conduct this study, the boundary lubrication test fixture was used in combination with the lubricant screening test fixture. The lubricant screening test fixture was first used to determine the relative performance of the different oils and additives. After the relative ranking of the lubricants was established, some of the oils were then tested in the accelerated life test fixture to see if the performance trends established by the screening tests were repeated in longer term tests.

The newer version of the lubricant screening test fixture, along with INA GT1 bearings, described earlier, was used to perform the screening tests. These tests were run at a speed of 1800 rpm and were conducted in vacuum at a base pressure of 10^{-7} torr. The reaction torque of the test bearing was monitored continuously; failure was defined to occur when

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the reaction torque of the test bearing exceeded a level 3x the initial run-in torque. For these tests, a 4 lb/ball load (288 ksi peak contact stress) was used in the test bearings, and each test bearing was lubricated with 60 μl of oil. This amount was somewhat excessive, but it was used to eliminate lubricant starvation as a failure mechanism. Table 2 lists the oils and oil/additive combinations that were tested in the lubricant screening test fixture.

While the screening tests were proceeding, the boundary lubrication test fixture was used to conduct longer term wear tests using the same oils and additives. The specimens in these tests were loaded to a stress level of 113 ksi and were run in an oscillatory manner with an amplitude of 0.10 in. and a frequency of 1 Hz. The tests were conducted in a vacuum environment for 1000 to 1500 hr. This time range was chosen because its length was considered sufficient for wear processes to occur.

The results of the screening tests appear in Figure 5. This figure clearly shows that oils formulated with wear additives outperformed the base stock oils. The poor performance of the unformulated Pennzane underscores the necessity of using boundary layer additives under these conditions. Of the combinations evaluated in the screening tests, Pennzane with TCP had the longest life. The addition of Pbnp to Pennzane also improved the life of the oil but not to the extent achieved by TCP. (See subsequent discussion on failure modes for the two additives.) NPT-4 was not formulated with any of the wear additives because its chemical structure and reactivity causes the formation of a protective boundary layer. However, the boundary layer produced by NPT-4 is not as effective as the boundary layer produced by the oil additives. Furthermore, the method by which NPT-4 forms its boundary layer (chemical reaction with the steel surface) may result in damage to the bearing surfaces.
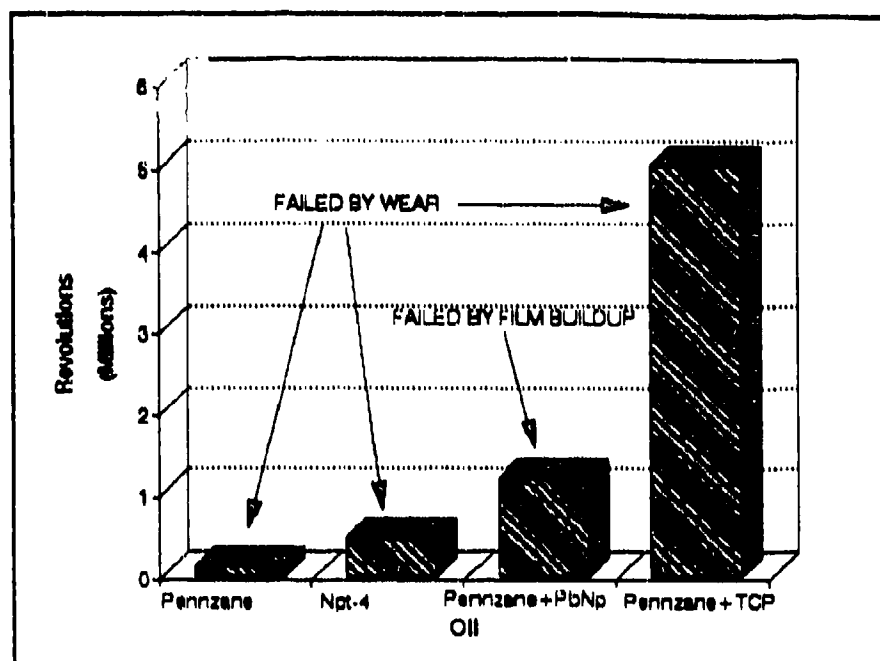
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Figure 5.   Screening test results (harmonic drive).

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Following these tests, the surfaces of the test specimens were analyzed with energy dispersive x-ray (EDX) and/or Auger electron spectroscopy (AES). The profiles of the surfaces were also measured using a DecTak 3030 profilometer. These analyses were performed to determine the failure mechanisms of the test specimens and to assess the performance of the boundary wear additives.

The posttest analyses revealed that the test bearings lubricated with TCP- and Pbnp-containing oils failed by different mechanisms. The bearings lubricated with Pennzane + TCP appear to have failed by a wear process that involves surface distress and metal removal from the wear track. Figure 6 contains two surface profile scans across the wear track of a Pennzane + TCP lubricated bearing. The graphs in this figure indicate that a significant wear trough, along with increased surface roughness, occurred in this test bearing. The bearing appears to have failed when the additive could no longer provide satisfactory boundary layer protection. The bearings lubricated with Pennzane + Pbnp, on the other hand, do not appear to have failed as a result of metallic wear. Instead, their increased friction appears to have been caused by the formation of a thick, lead-containing carbon film on the wear surface of the disk. The evidence for the existence of the carbon-lead film was first detected by EDX spectroscopy and was later confirmed by profilometry measurements. The data are consistent with other studies of the reactivity of Pbnp reported elsewhere.[5-7] Figure 7 contains several EDX scans of a test sample containing a carbon-lead film. As the plots show, only the scan of the regions within the wear zone contain elemental carbon and lead in any quantity. Outside of this area, there is little evidence of these elements on the surface of the sample. Figure 8 shows profilometry traces taken across the wear track of the same
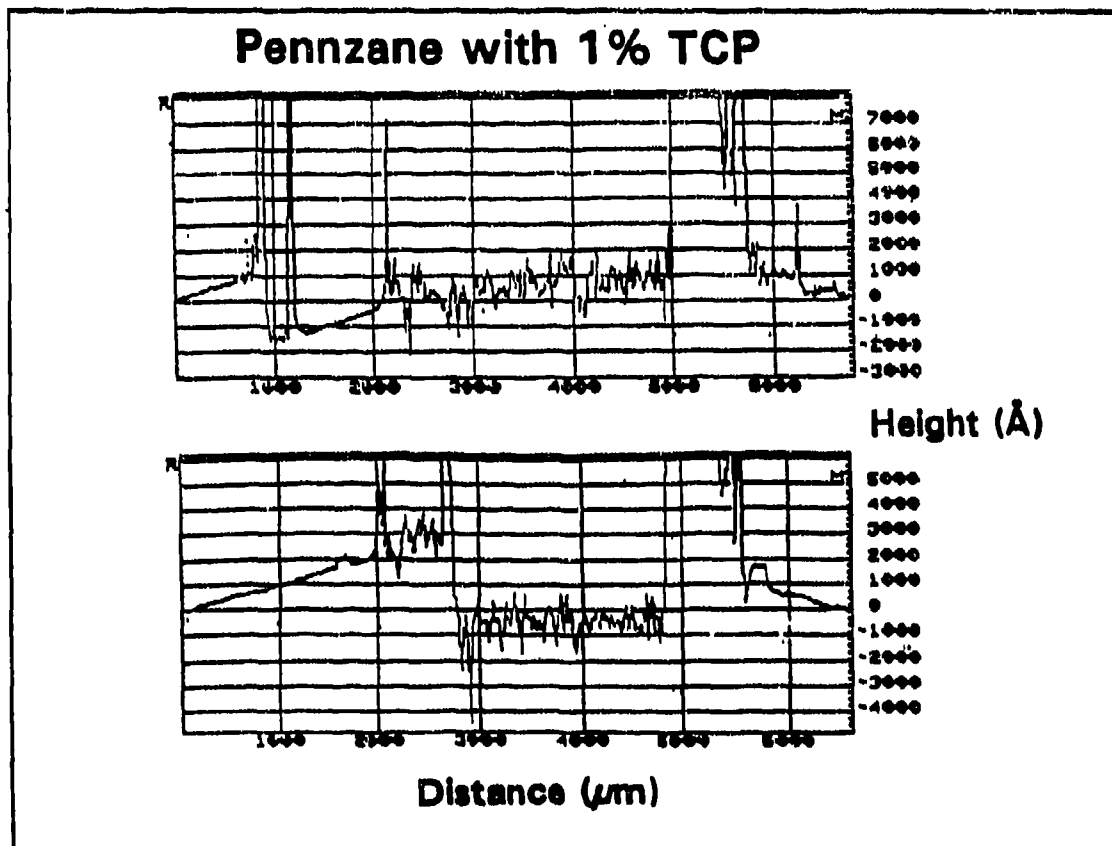
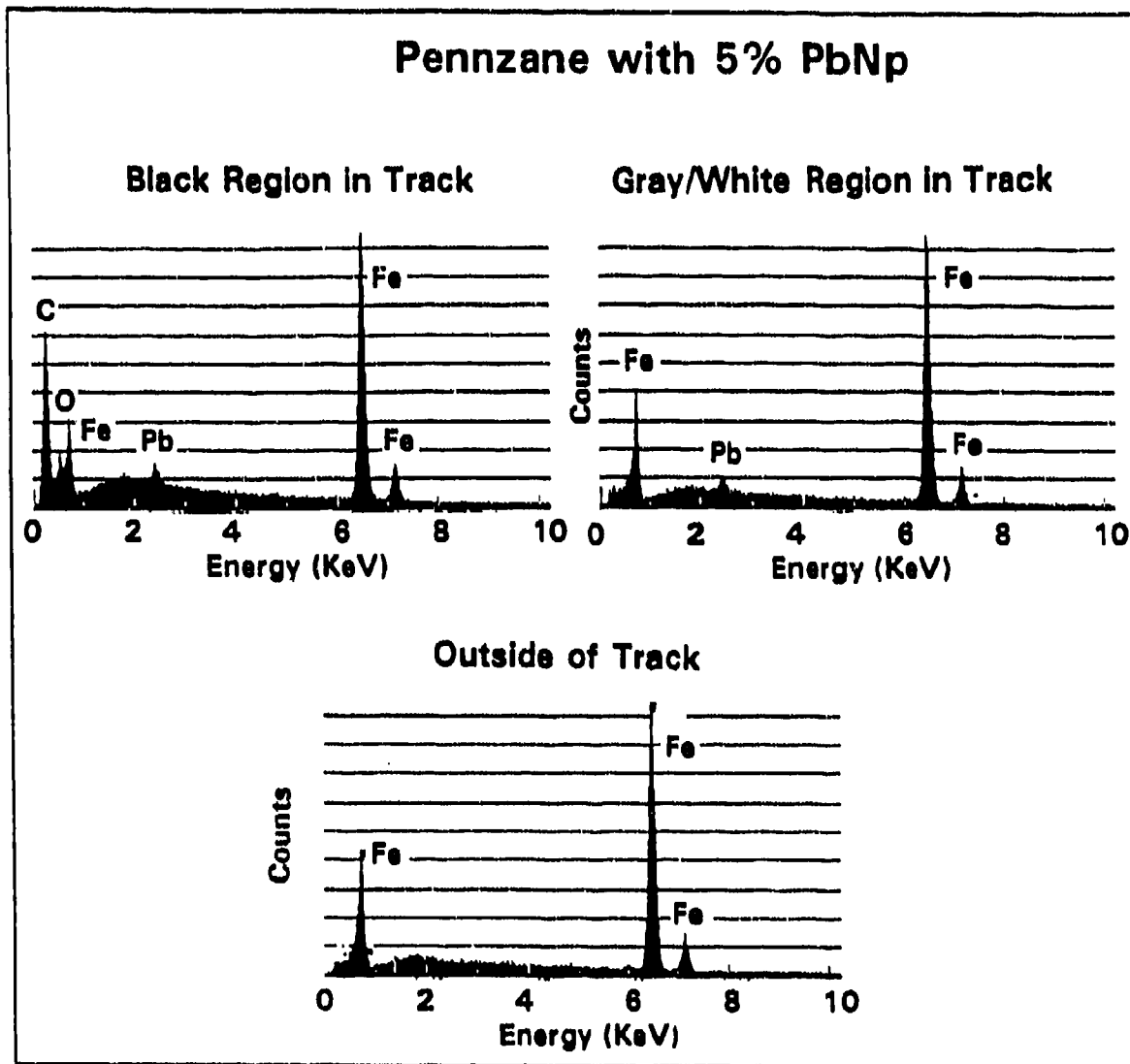22

Figure 6. Profiles of Pennzane + TCP wear disk.

Figure 7.   EDX spectra of Pennzane + Pbnp wear disk.
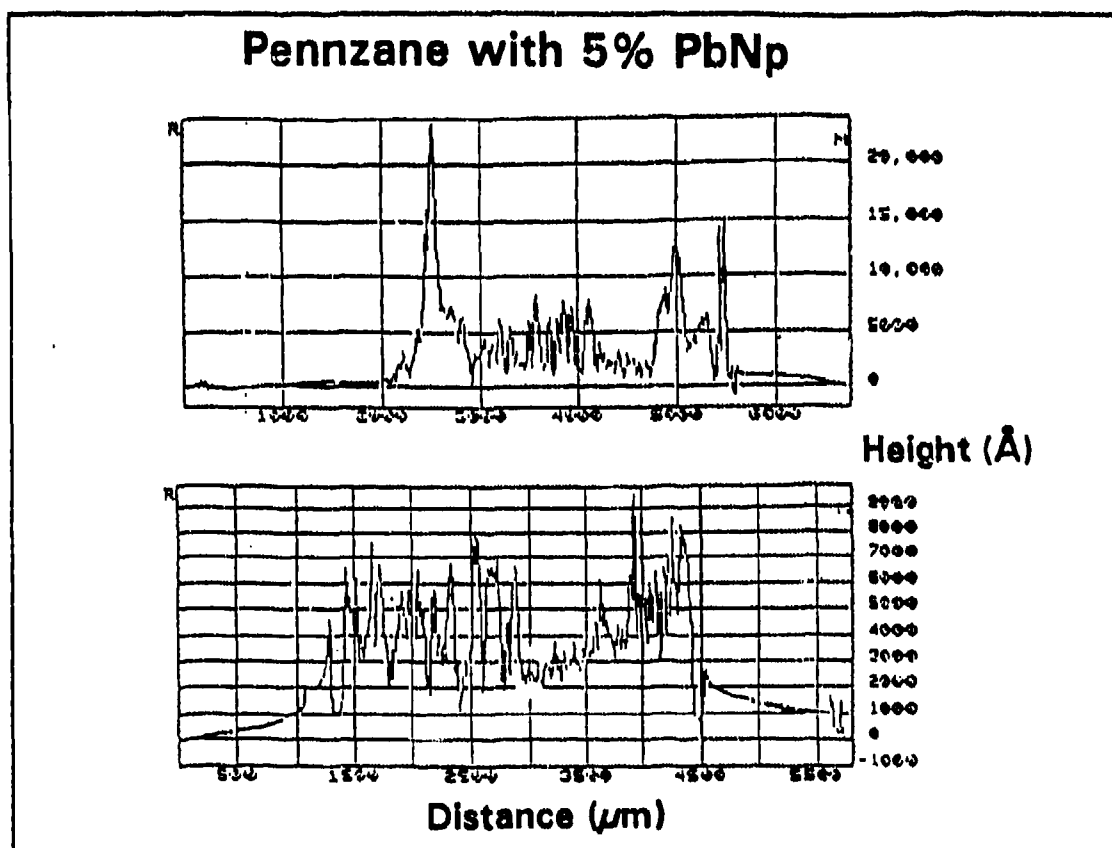
Pennzane with 5% PbNp

Figure 8.   Profiles of Pennzane + Pbnp wear disk.

raceway. The graphs confirm the existence of a film buildup across the wear track and indicate that the film is between 0.1 and 0.5 μm in thickness.

The apparent differences in the performances of TCP and Pbnp in Pennzane have suggested that the two additives cannot be directly compared to each other. The test results suggest that TCP and Pbnp have different boundary protection modes. Pbnp appears to function by generating a relatively thick film between the contacting surfaces. This characteristic makes Pbnp ideally suited to high stress applications where maximum protection against wear is desired and increased friction is not a factor. TCP, on the other hand, seems to function by reacting with the steel and forming a thin, friction-reducing film on the contacting surfaces. These characteristics make TCP well suited to light or intermediate loads, where low friction and long life are desired. However, as shown by Figure 6, TCP does not provide the same degree of protection against wear as does Pbnp. This is an important consideration, especially in situations where bearing stiffness is crucial to the success of the application.

These results illustrated some of the difficulties of performing comparative screening tests on different lubricants and the need for careful consideration of the end application in making final selections. Many factors, such as load, speed, temperature, and material compatibility must be evaluated when determining the test conditions. Posttest analyses of the test components are essential to proper interpretation of the test results. Posttest analysis was extremely useful in our case, because it allowed us to reevaluate the ranking established solely by the screening test wear lives. Based on the screening test wear lives, Pennzane with TCP appeared to be the best choice. However, for the purpose of the harmonic drive application, the low friction produced by TCP is not as important as the improved

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wear protection provided by Pbnp. Based on bearing wear as the ranking factor, Pennzane with Pbnp is the best lubricant combination tested.

Due to the length of time necessary to perform the accelerated life tests, not all of the lubricant combinations tested in the screening tests were tested in the boundary lubrication test fixture. Nevertheless, the results obtained from this test fixture have confirmed most of the findings of the screening tests. Posttest analyses, similar to those performed on the screening test samples, have given us sufficient data to make this conclusion. Table 3 is a compilation of all of the test results from the screening tests and the low speed oscillatory tests. From these data, it is evident that the lubricant behavior in the screening tests has been replicated in the accelerated life tests. In addition to our findings, life testing performed on actual harmonic drive mechanisms with NPT-4 and Pennzane + Pbnp has confirmed these results. Additional tests with Pennzane and Pbnp have been planned. These tests will investigate the effects of varying additive concentrations on wear performance and will look for an optimum additive concentration.

C. CASE 3 - REACTION WHEEL SUPPORT BEARINGS

The lubricant screening test fixture was also used to evaluate lubricants that have been considered for use in reaction wheel support bearings. In this study, the fixture was used to test a well known spacecraft lubricant, SRG-40, which is a highly refined mineral oil, and two synthetic oils that were considered as replacements for SRG-40. The synthetic oils consist of NYE 179, a PAO oil, and NYE UC-7, which is a poly-ol-ester oil (POE). All of the oils tested were formulated with TCP. Table 4 lists some of the properties of the oils tested.

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Table 3. Test Results of Harmonic Drive Case Study

Oil	EDX/AES		Profilometry	
	Screening tests	Accelerated life tests	Screening tests	Accelerated life tests
NPT-4	carbon film	carbon film	N.A.	wear patch
Pennzane	no film	no film	wear trough	wear patch
Pennzane + Pbnp (5%)	carbon + lead film	carbon + lead film	film buildup	film buildup
PENNZANE + TCP (1%)	thin carbon	N.A.	wear trough	N.A.

Table 4. Oils Tested in RWA Case Study

Oil	Viscosity (300 K)	Pour point	Vapor pressure (300 K)
SRG-40	30.3 cs	-26 °C	~ 1×10^{-4} torr
NYE 179 (PAO)	40.8 cs	<-60 °C	~ 1×10^{-7} torr
NYE UC-7 (POE)	45.8 cs	-57 °C	~ 1×10^{-7} torr

Reaction wheel assemblies (RWAs) and gyroscopes often operate in atmospheres of helium or hydrogen gases. The particular RWA of interest uses a He-O_2 mixture (P_{ux} = 0.5 atm with 2% O_2), so it was decided to perform lubricant screening tests in several different He-O_2 environments. These environments consisted of: 10^{-7} torr vacuum simulating a worst-case scenario of an on-orbit leak; one-half atmosphere of 98% He 2% O_2, the operating environment used on orbit; and one-half atmosphere of pure He. The last environment was chosen to see if the inclusion of oxygen was really necessary for the antiwear additive TCP to function. Many spacecraft designers have included oxygen in the fill gases of reaction wheels based on intuition rather than experimentation. The premise for including oxygen has been that TCP will only bond to an oxide surface. Hence, the addition of oxygen to the helium fill gas will ensure that all metal surfaces will be covered with an iron oxide film, which can be replenished if worn off. In our experience, however, we have never observed that oxygen is necessary for TCP to function correctly. In fact, our concern in this study was that the inclusion of oxygen in the fill gas might actually be detrimental to life of the bearing because it could degrade the lubricant. Therefore, several additional tests were run in a 7 torr atmosphere of O_2. This amount of O_2 represents the equivalent of the 2% O_2 added to the helium. It was felt that this operating environment would accentuate both the reactive effect of oxygen on the lubricant and any potential effect of oxygen on TCP.

The test conditions and procedures for the screening tests in this study were virtually identical to those performed for the harmonic drive actuator. The only difference, aside from the different environments, was that

some of the tests were run with a lighter applied load (3 lb/ball).

The wear lives of the screening tests performed in vacuum and 7 torr O_2 appear in Figure 9. In this figure, the wear lives represent the mean values of several tests per oil. As the graph indicates, the synthetic oils outlasted SRG-40 by a wide margin in vacuum. The main factor that accounts for this result seems to be the high vapor pressure of SRG-40. It is felt that this high vapor pressure, combined with the extreme operating environment, led to rapid lubricant loss from the bearings and subsequent failure. The synthetic oils, with their lower vapor pressures, appear to have remained in the bearings for a greater duration, which resulted in their longer wear lives. Gas chromatography was later performed on the oil residues of all the test specimens, which confirmed that there was significant lubricant evaporation in the SRG-40 test bearings, while there was much less evaporation in those lubricated with the synthetic oils.

Although not all of the oils were tested in 7 torr of oxygen, the results from those that were tested indicate that the presence of oxygen in the operating environment did not improve the test lives, and, in the case of UC-7, the wear life was significantly decreased. Figure 10 is a plot that compares the bearing reaction torques for UC-7 in vacuum and in the oxygen environment. From this figure, it is obvious that the addition of oxygen to the operating environment results in higher torque, more torque noise, and greatly reduced wear life. Visual inspection of the bearings after the tests also revealed extensive lubricant degradation in the oxygen tests and little or no lubricant breakdown in the vacuum tests. This finding was later supported by findings from the one-half atmosphere tests.
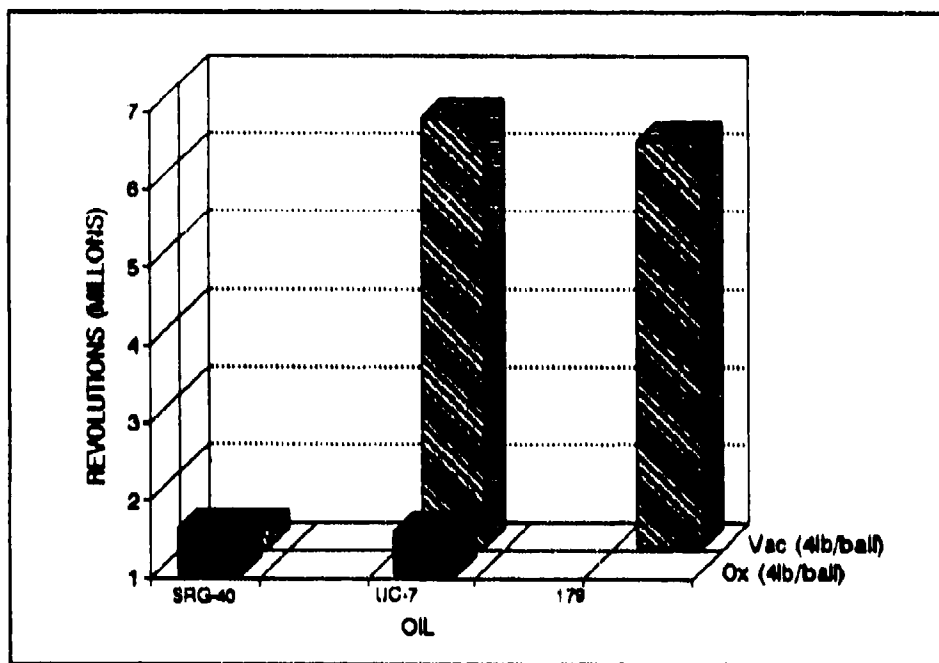
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Figure 9.   Vacuum and O_2 screening test results (RWA).
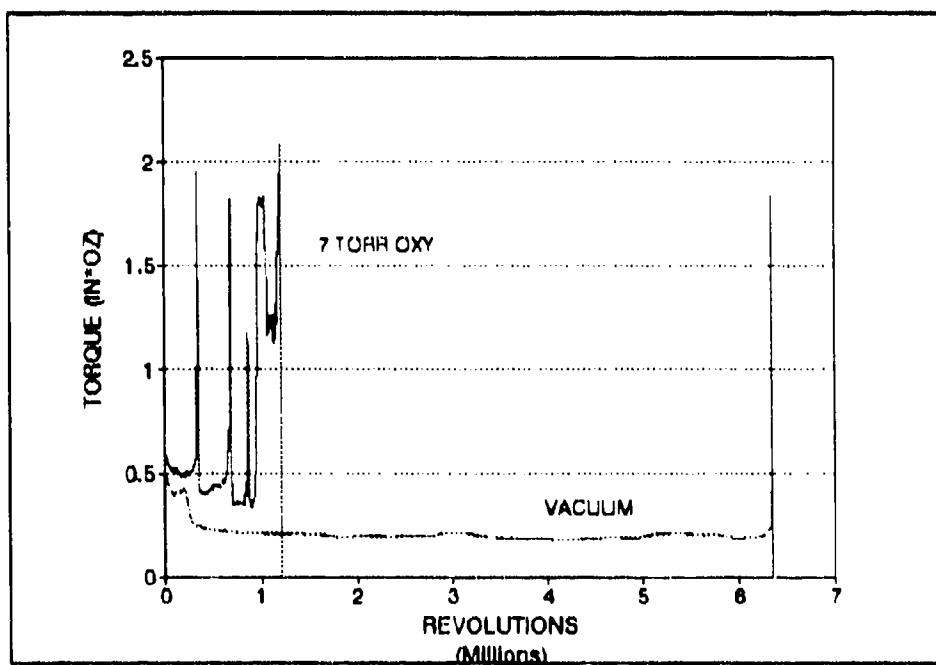

Figure 10.   Torque traces of UC-7 in vacuum and O_2 (RWA).

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The wear lives from the tests performed in the one-half atmosphere environments appear in Figure 11. In general, these tests ran for a much longer period because the presence of the fill-gas reduced lubricant evaporation and allowed the bearings to operate at a lower temperature, providing a more direct comparison of the respective boundary layer performances of the oils. The tests that were performed with the helium-oxygen mixture (SGR-40 only) confirmed the results obtained from the 7 torr oxygen tests. The test lives in this case were significantly shorter than any of those performed under a helium-only environment. In the helium-only environment, UC-7 was the only oil that failed consistently, according to our torque failure criteria. In contrast, the tests performed with the other oils were either terminated before failure, due to time constraints, or were run for a great length of time in order to fail (NYE 179). Consequently, posttest analyses, similar to those performed in the harmonic drive study, were carried out on test specimens from all of the test environments to clarify the results.

The most notable finding from the surface analyses (AES and profilometry) is that TCP does not appear to function as an antiwear additive in UC-7. The Auger spectroscopy performed on the UC-7 wear disks never detected any more than a minute trace of phosphorus on the steel surfaces. The absence of phosphorus in this case, combined with the detection of a carbon residue on these surfaces, suggests that the reactive nature of ester oils interferes with the normal protective mechanism of TCP. In contrast, all of the samples run with NYE 179 and SRG-40 contained phosphorus in the wear track, indicating that TCP was active in this region and that oxygen is not required for TCP to function.

The profilometry measurements revealed large wear troughs in the wear tracks of the UC-7 test specimens. Measurements made of the other wear disks indicate that SRG-40 operates
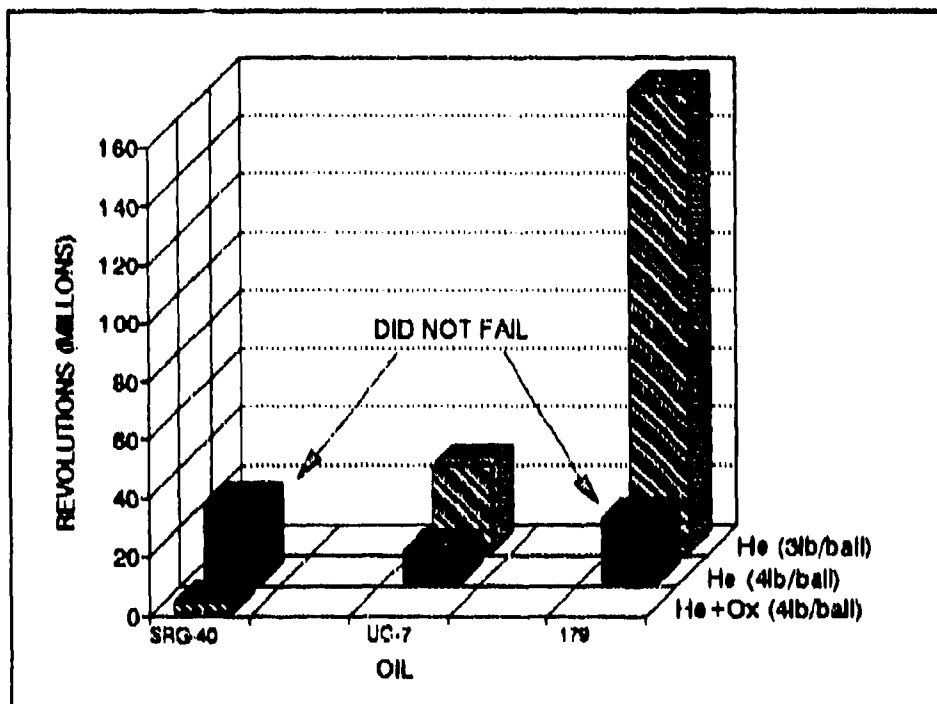
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Figure 11. One-half atmosphere screening test results (RWA).

with less wear than UC-7, and that NYE 179 appears to have the lowest wear rate of all the oils tested under these conditions. The presence of this type of wear and its relative absence for the other oils suggest that the reactive boundary layer film, which UC-7 generates under use, is not as effective as TCP in preventing wear. In fact, this method of boundary layer protection may actually damage the steel surfaces of the bearings and cause increased "chemical" wear.

The wear lives of the screening tests, and the posttest analyses performed on the test specimens, indicate that Nye 179 was the best overall choice for this application. Its performance in vacuum was far better than that of SRG-40, and its performance in helium was considered to be the best of all the oils tested. Furthermore, the use of oxygen as a component in the fill gas of reaction wheels was determined to be unnecessary and generally harmful to the life of the bearing lubricant. These findings have been conveyed to the manufacturer and have been used to determine the configuration of a reaction-wheel-lubricant validation test that is currently in progress.

IV. SUMMARY

The lubricant screening test fixture and the procedure
developed for its use have proven to be quite useful for
acquiring performance information for spacecraft lubricants.
The chief advantage of this approach is the ability to obtain
qualitative rankings of different lubricants through the use
of low cost, short duration bearing tests. This advantage is
significant, when considering the lengthy test times and high
costs associated with most spacecraft lubricant tests. The
test fixture and test procedure also can be used to perform
more fundamental studies on the interactions of lubricants and
additives with bearing surfaces. The case studies described
in this report have demonstrated that this test fixture, in
conjunction with the appropriate posttest analyses, can be
used to identify different wear protection mechanisms as well
as to determine bearing/lubricant chemistry.

The results from the case studies, however, have also
demonstrated some of the limitations of this approach. The
results have shown that screening tests alone are not
sufficient for most lubricant studies. Because the test
bearings used in these tests are significantly different from
the configuration of most satellite bearings, it is almost
impossible to predict accurately an application's life based
solely on screening test lives. The test results from the
case studies have also highlighted the necessity of choosing
the appropriate test conditions. The effects of test
parameters such as load, speed, temperature, atmosphere and
lubricant quantity on the test life need to be understood if a
valid lubricant ranking is to be established. Even more
important is the selection of appropriate posttest analyses.
These analyses are often needed in order to identify
performance traits, such as additive surface effects, which
may not be discernible in a torque trace or a wear life. This

need was most evident in the harmonic drive study, where posttest analysis rearranged the initial lubricant ranking based on wear lives.

If these limitations are understood, however, the proper use of lubricant screening tests can play a major role in the process of qualifying a spacecraft lubricant. The tests can save considerable time and expense by identifying unsuitable lubricants, thus eliminating unnecessary life tests. In addition, they can often be used to troubleshoot lubricant problems with existing mechanisms, as was most noticeable in the case study involving the reaction wheel lubricants. In this case study, with the screening tests, we were able to identify almost all of the lubrication problems associated with the actual flight bearings. Considering their strong points, it seems reasonable to integrate lubricant screening tests into the overall process of lubricant flight qualification.

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